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Otto Regner
5491 Winchester
Troy MI 48085

Mr. Dirk Wright
Primary Examiner
Art Unit 3681
Assistant Commissioner for Patents
Washington, D.C. 20231

Re: Application / Control Number 10/042,626
Art Unit: 3681

Dear Mr. Wright:

Accompanying this letter are the reworked drawings and description of my patent application documents composed of the following:

- Seven pages of revised drawings for the cyclo patent application.
- Two pages of performance charts showing real-world cyclo behavior.
- Seven pages describing the patent features, figures, tables and the sixteen claims.

I am sorry for the delay in answering your concern about the above patent application. I had a total of two computer and hard drive failures and was only able to recover part of the patent drawings and data, even with specialized technical support. In addition, it was very difficult to get documentation important to this patent. The documentation is in the accompanying Charts 1 and 2.

The charts were made from machines equipped with cyclo drives cited by Mr. Wright in his letter/report from February 11, 2003. It is important to know that I worked with these companies in Japan, Europe, and here in the Unities States.

I invented the three disk, three cam individually spaced cyclo drive in 1982 when I was employed by General Motors Corporation in Warren Michigan. At that time, I was not required to sign over any patents to my employer.

Regretfully in 1986, when I finally had my patentable design realized, I was fired within two minutes of testing and verifying its performance and usefulness. After that time, I realized that there were still some shortcomings in the performance of the cyclo gears which I have addressed and eliminated with this patent Application #

10/042,626. The differences between drawings sent to me and improvements I made are as follows:

A) The important cyclo tooth and force contact is now closer to the spur gear standard angle of 20 degrees. The cyclo tooth spacing densities shown in the noted samples sent to me are not even close to my patent and performance. In addition, because of the high compound forces needed and generated by the other gears' contact angle, there is much more vibration generated and a much lower efficiency found than with my design. I found no infringement, however, in the cyclo tooth design or its spacing or its tooth depth. The all important efficient cyclo relations, however, are missing and are nowhere sited. This is the important basis of my invention.
B) The eccentric's size and throughputs are much larger in my design.
C) There are seven through-puts compared to only one sited.
D) There is not one slim line, pancake-type cyclo axis sited except in my patent.
E) There are only two twist-free cyclo axes shown in the patent by Otto Regner, one on Figure 1 and one more in Figure 2.
F) The usefulness and mounting friendliness is much greater with more taped screw holes.
G) The damping of a servo cyclo axis is guaranteed with the design shown which includes large supported axes and shafts.
H) With the single directly mounted absolute encoder disk, vibration is practically impossible.
I) The damping filter filed herein will reduce electronic shortcomings in the servo control.
J) The geometrical relations shown in Table 2 will maximize the features of these designs and cyclo gear axes for many centuries to come.

Again, Mr. Wright, I apologize for the delay and hope the revised application meets your expectations and approval. Should you have any questions or would like to have something altered to meet your standards, or if there are any added fees, please let me know by calling or faxing to 248-828-9688 or by sending e-mail to flextec@wowway.com.

With high regards,

Otto Regner   12-8-2003

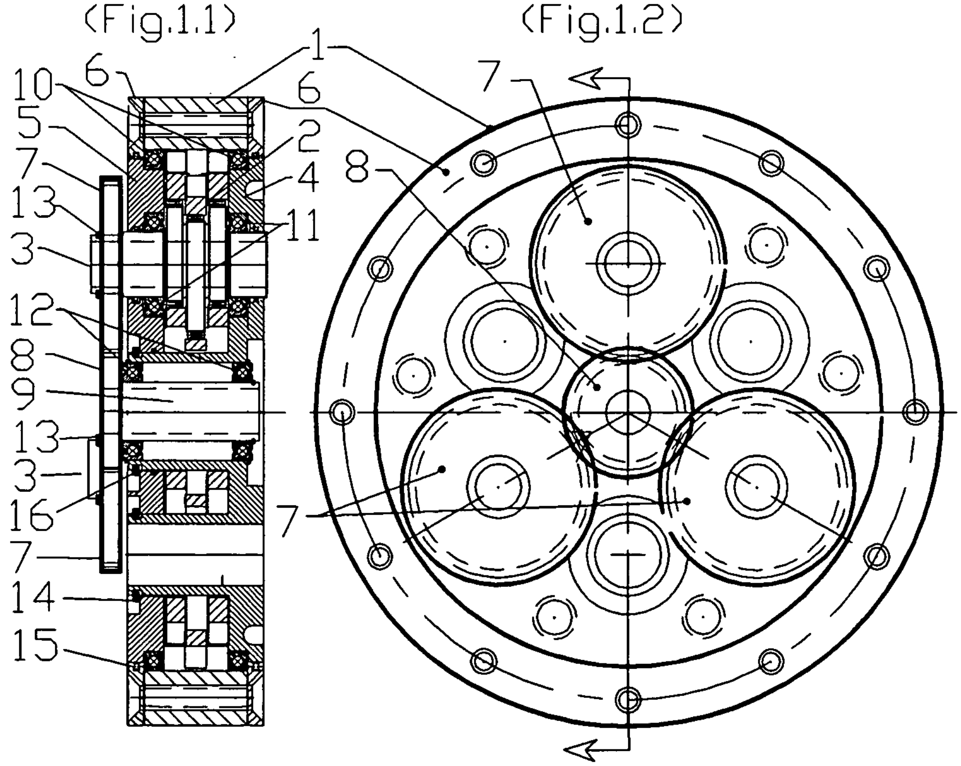
Otto Regner

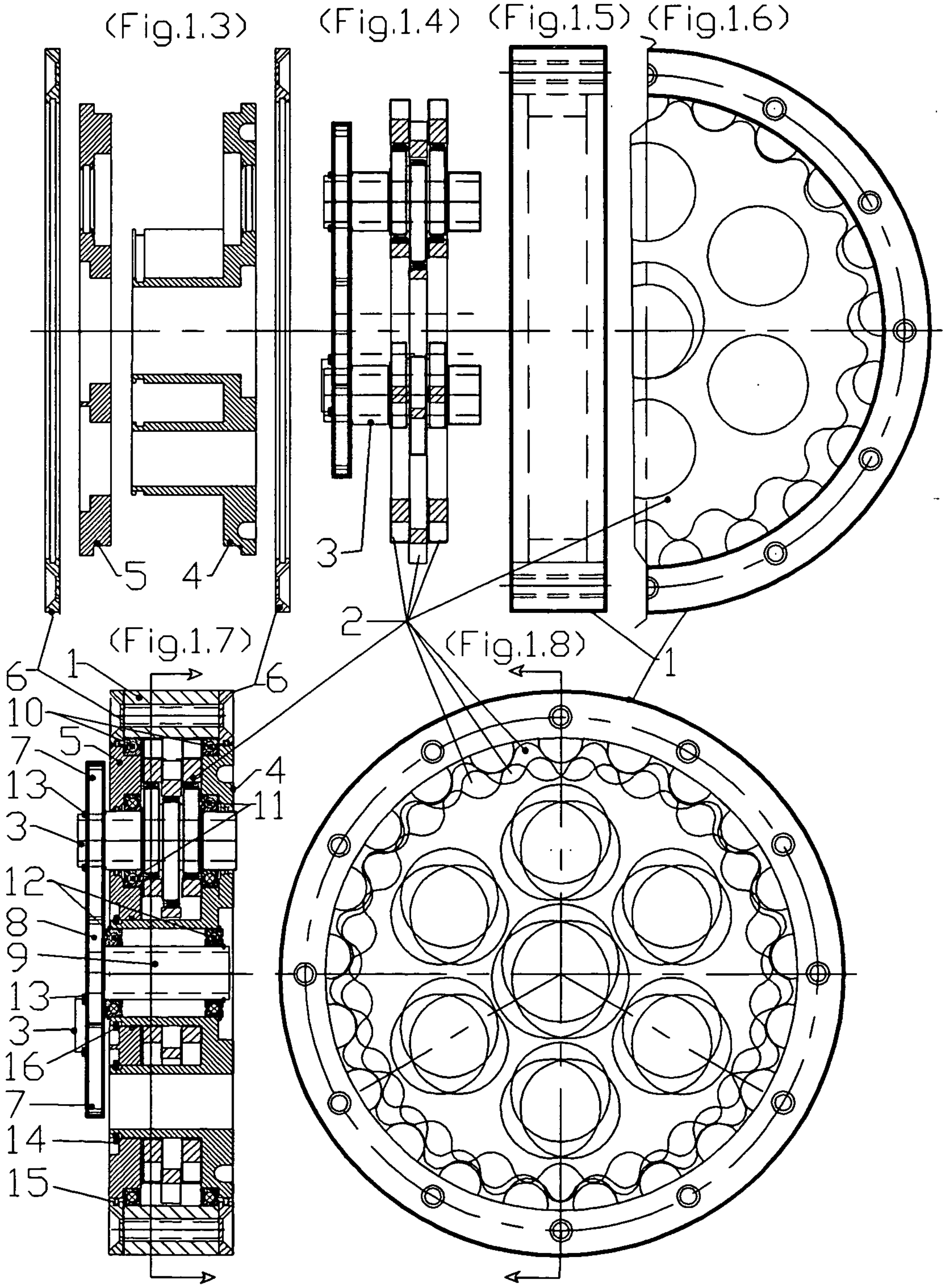
Fig.1

Table 1

1 CYCLO ID-GEAR HOUSING
2 CYCLO OD-GEAR DISK
3 ECCENTRIC 0+120+240deg. HOLLOW SHAFT
4 DRIVE-THROUGH HOLLOW FLANGE
5 CONTAINING FLANGE
6 BEARING RETAINER
7 PLANET GEARS
8 PLANET SUN GEAR
9 SUN GEAR HOLLOW AXIS
10 BEARING CYCLO AXIS
11 ECCENTRIC BEARING
12 BEARING SUN GEAR SHAFT
13 SNAP RING PLANET GEAR
14 SNAP RING FLANGE
15 SEAL X-TYPE
16 SNAP RING CENTER OF HOUSING

(Fig.1.1) (Fig.1.2)





10/042,626 Cyclo Patent Otto Regner Page 2 Dwg.

Table 2

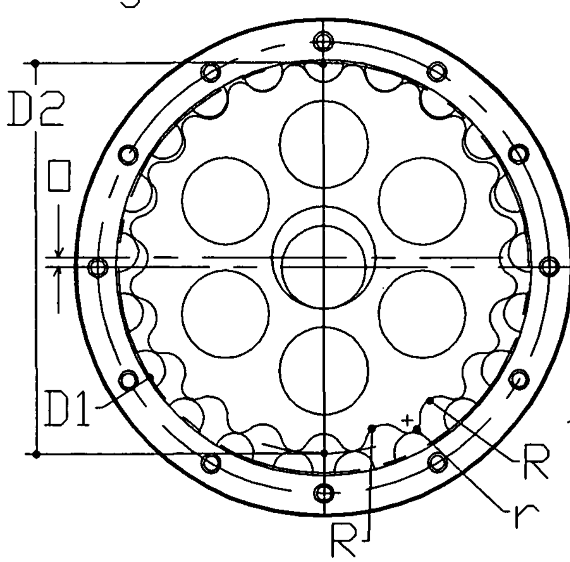
Cyclo Gear Relations and Symbols:

R = radius of cyclo tooth
r = r of Arc Tan(R, $D_{1,2}$, R)
D = diameter at tooth centers
□ = offset of eccentrics
Z1 = number of cyclo gear teeth
Z2 = number of cyclo disk teeth

Relations:

Z2 = Z1 - 1
D1 = Z1 x R
D2 = Z2 x R
□ = R/2
U cyclo = R2/R1 -1
U total = (Z sun / Z planet) (U cycle)
e = Ecc. Index = 360deg /No of Cyclo Disks

(Fig.1.9)



(Fig.1.10)

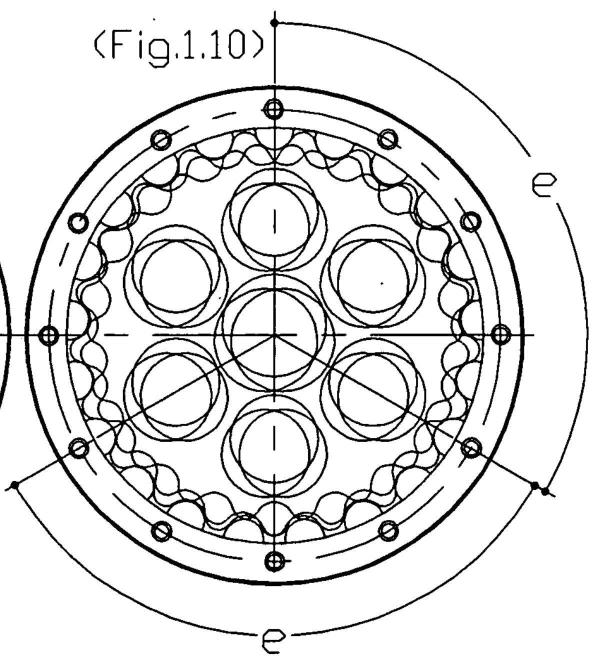


TABLE 3

	R	Z	D
Sample Cyclo	0.1	3	0.3
Gear Relations	0.1	4	0.4
From 3 to 11	0.1	10	1.000
+ 60 and 61	0.1	11	1.100
Cyclo Teeth	0.1	60	6.000
(Fig. 1.11)	0.1	61	6.100

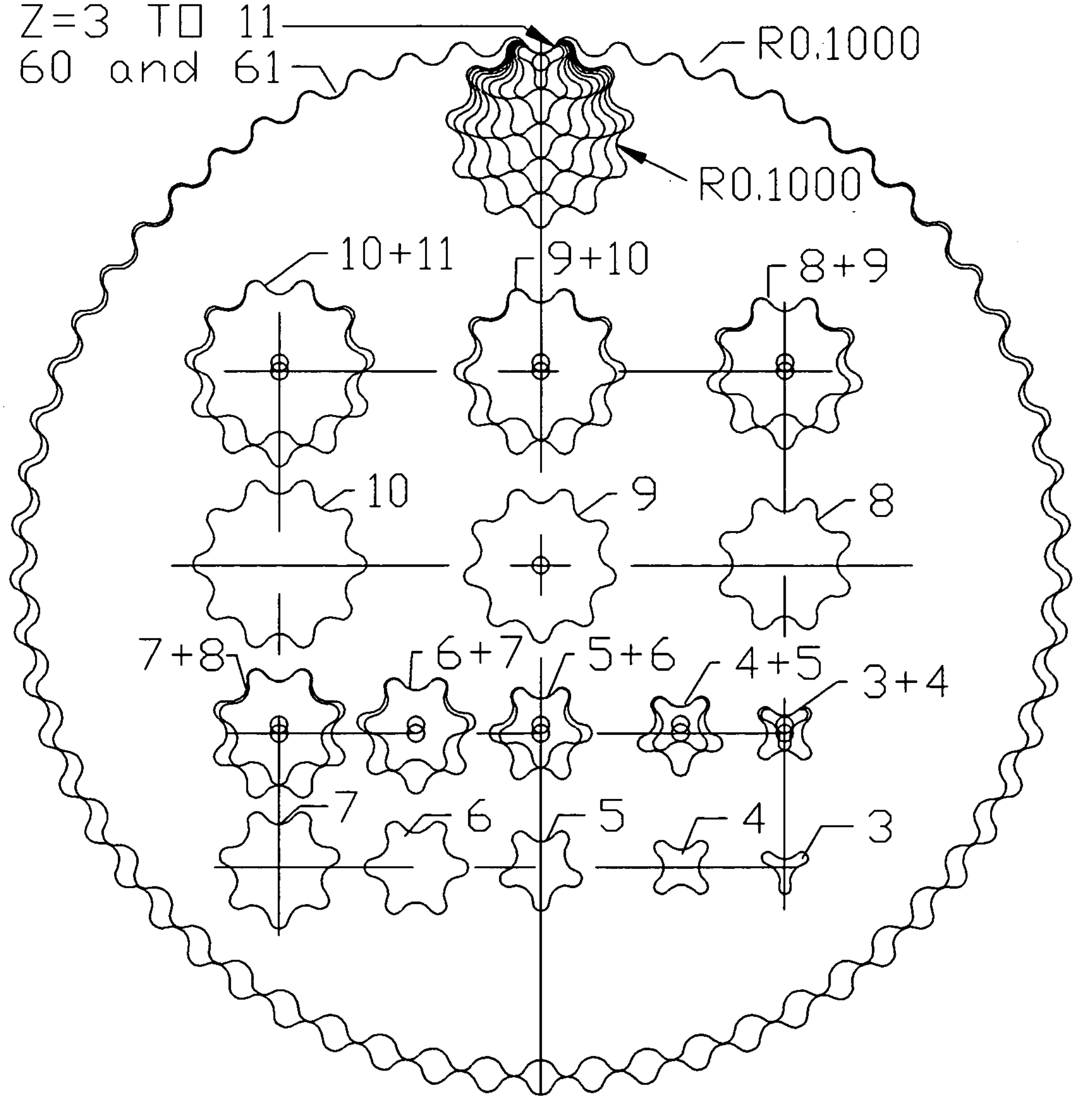
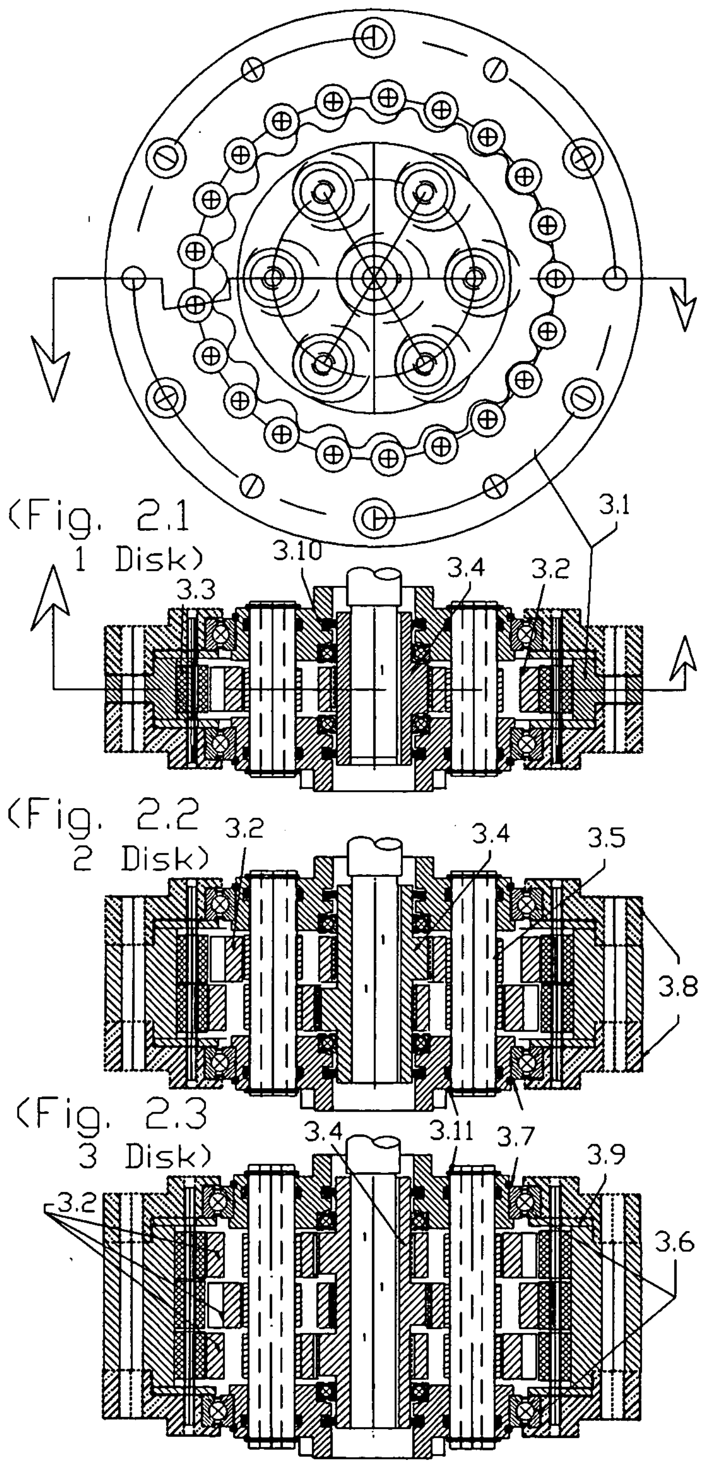


Fig. 2



Center-Driven Cyclo Gear Axes with one, two or three Wave Disks with six Drive-Out Pins and Low-Friction Bushings.

TABLE 4

3.1 Cyclo Gear
3.2 Cyclo Disk(s)
3.3 Cyclo Rollers
3.4 Eccentric(s)
3.5 Hollow Pins
3.6 Bearing Flg.
3.7 Snap Ring
3.8 End Covers
3.9 Stop Rings
3.10 Shaft Seal
3.11 Snap Ring

Fig. 3

FREQUENCY SHIFT AND SERVO FILTER TO CONTROL CRITICAL FREQUENCY VIBRATION

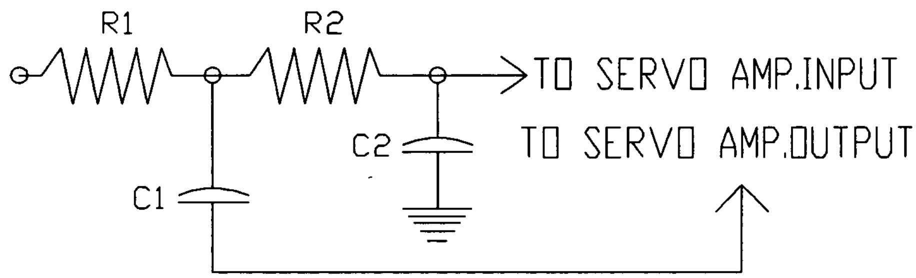


Fig. 4

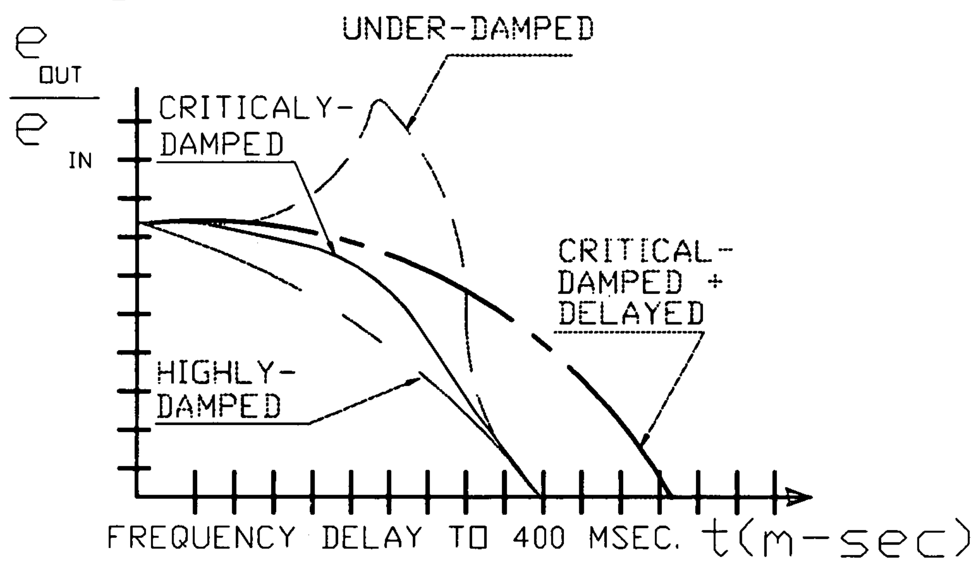


Fig. 5

ONE DISK ABSOLUTE ANGULAR ROTATION ENCODER USING LOW-POWER INFRARED LED, TTL UP/DOWN COUNTER WITH SHIFT REGISTER AND LOCAL RECHARGEABLE BATTERY POWER BACKUP

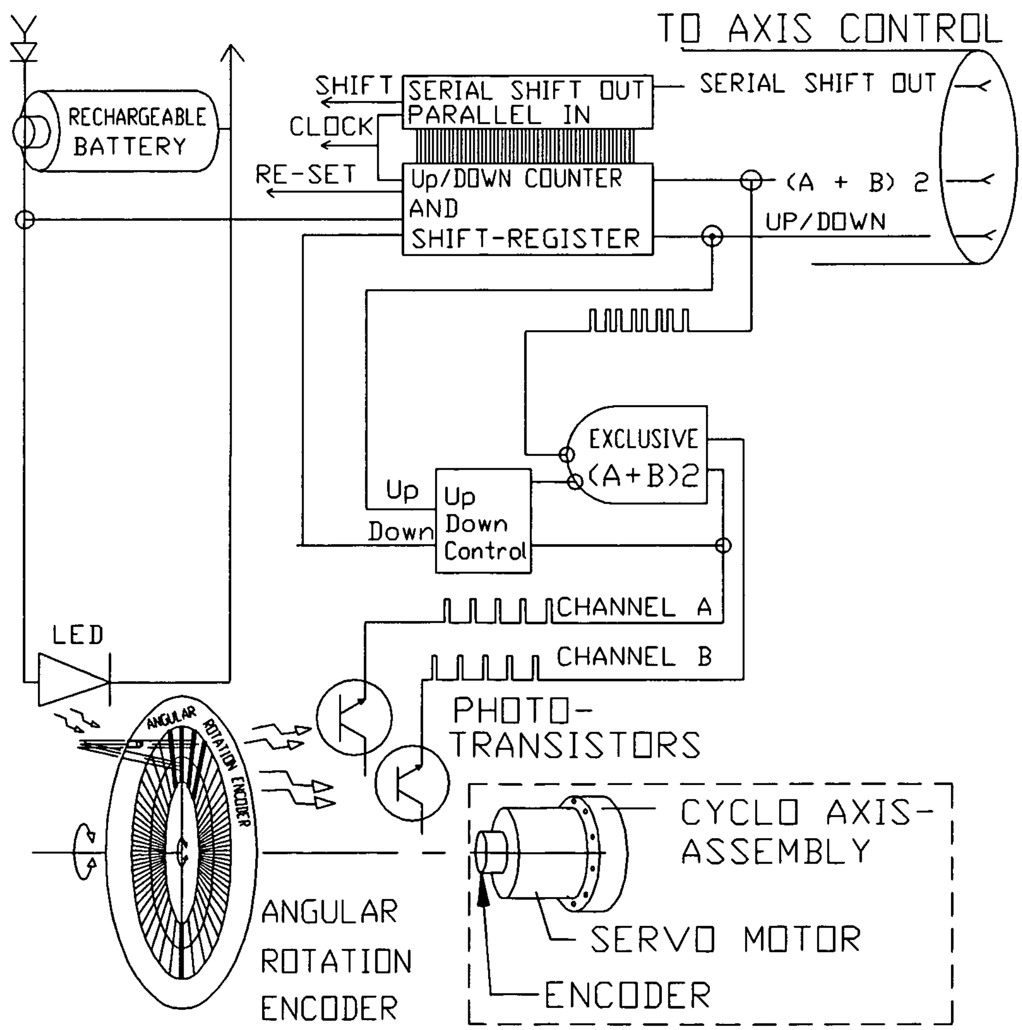


Chart 2 of Cyclo Patent for Otto Regner

The chart below is from the S-400 Serial # F1741 Fanuc (GMF-Robotics) end-effecter movements. The robot's gears are cyclo torque multipliers and the Theta Axis was only programmed to move for charting.
The programmed time was over 20 seconds at a 10mm/second speed. The ABS vector was charted 5 seconds after start and for 10 seconds.
The produced chart tells us: there are 16 ABS (absolute) velocity changes larger than 30 mm/sec; the largest is 62 mm/second. The majority of 65+ velocity changes are in the range of 20 mm/sec. The main vibration frequency is 1.6 hertz and the secondary multiple of frequencies is 8 hertz. The critical base frequency is therefore 1.6 hertz with an "N" resonance factor of 5; 5 x 1.6 = 8 hertz, absolute.

ROBOT CHECK　　BRIEF OVERVIEW　　VELOCITY (mm/s)
Selepine　　PAGE:
Software group

DATE: 23-AUG-1985 14:52:32
LABEL: S-400; F1741; THETA AXIS VIBRATION; SPEED=10 MM/SEC

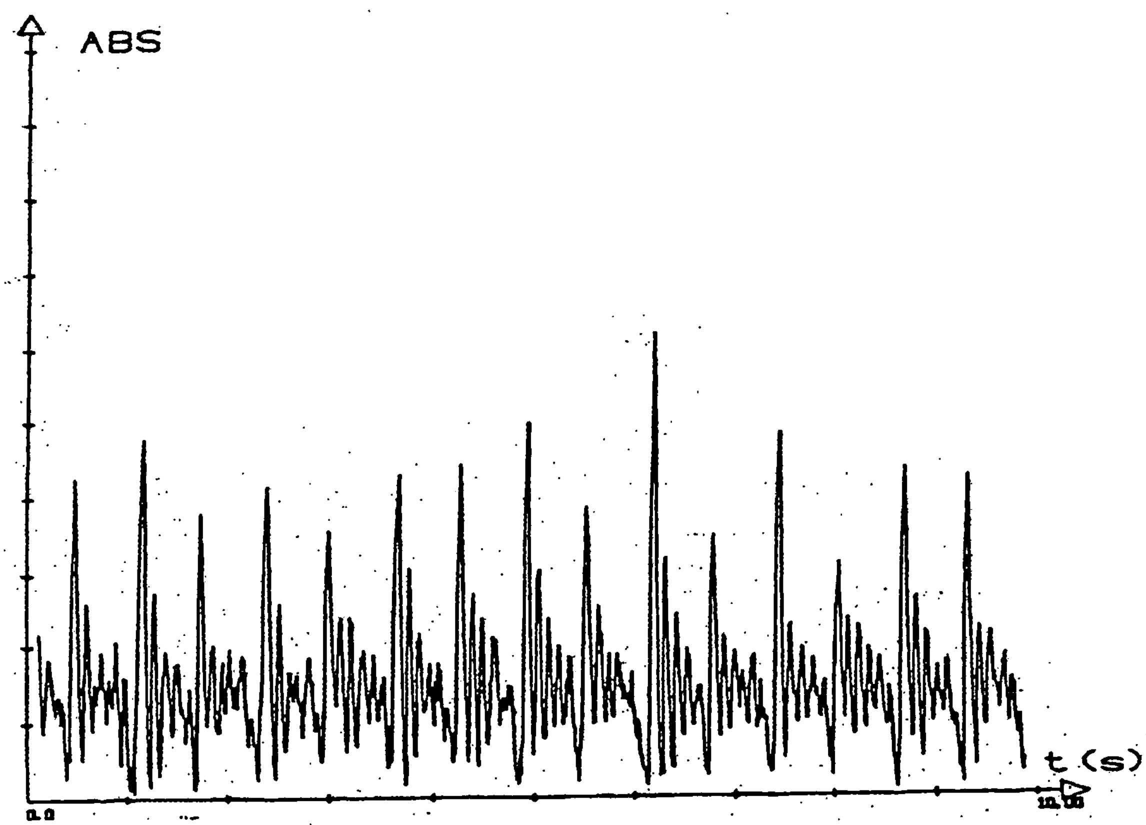
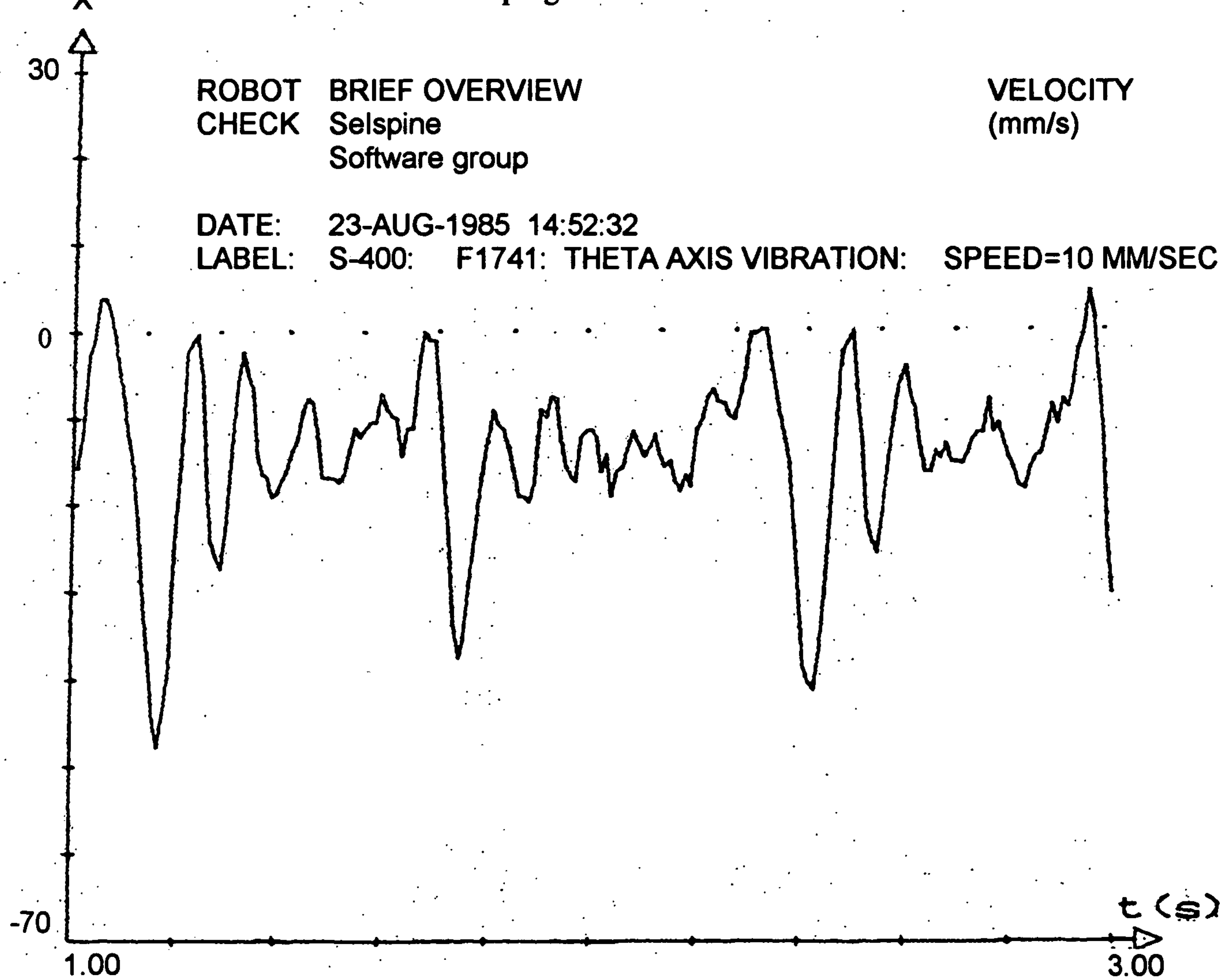


Chart 1 of Cyclo Patent for Otto Regner

The chart below shows the true S-400 Serial # F1741 Fanuc (GMF-Robot) end-effecter movements. The robot was equipped with cyclo torque multipliers. The base or Theta Axis was moving.

The measured movement was 2 seconds between start and stop. The vibration amplitude shown is -48 mm/sec to 5 mm/sec and basic 1.6 hertz and imposed multiple of 8 hertz at a constant programmed velocity of -10 mm/sec. The vibration movement in the X-plan is about three times that of the programmed movement.



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PATENT APPLICATION SUMMARY FOR CYCLO TORQUE MULTIPLIERS AND CONTROLS

Background of the Invention

This invention addresses geometric relationships and designs and the problems that occur with cyclo torque multipliers. Some of those problems are harmonic vibrations (See attached Charts 1 and 2.) and insufficient rigidity which are eliminated with the features described in this patent.

Drawings in Figure 1 (1.1 through 1.11) and Figure 2 (2.1, 2.2, 2.3) show that if the geometric design relations noted in Table 2 are used, a much higher mechanical rigidity, longer gear life, and easier use and application of cyclo gears is accomplished. The most obvious feature is a deeper tooth engagement. The result is a direct force contact. These designs incorporate cyclo-bearing hub-axes with hollow shafts and hollow torque pins. These inventions simplify the building of machines, particularly when multiple cyclo axes are used in sequence, as with base cyclo turn tables and waste and cyclo arm-wrist assembly clusters for robots and other frequently-linked tools. With these cyclo inventions, a more direct and deeper, three-vector multi-line engagement of the cyclo components is guaranteed. Therefore, variable load vectors are controlled and neutralized, and they will not generate harmonic vibrations.

Figure 3 depicts an electrical circuit that is a smoothening, anti-oscillating add-on filter for servo systems. Attached Charts 1 and 2 show oscillation and vibrations that can be reduced electronically with this smoothening circuit by applying this patented control technology whose results are explained and shown graphically in Figure 4.

The Figure 5 schematic shows the invention of a constantly powered absolute encoder system for controlling a complete servo cyclo axis. It is a single two-channel disk encoder with quadruple (exclusive "or" gate) up/down counter that is both backlash free and directly coupled to motor and cyclo axis, as well as to the program controller - computer.

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1, Table 1, lists the key parts of the three-disk cyclo gear axis invention. The accompanying drawing, Figure1.1, explains the basic design of the cyclo gear axis in cut view through the center of the assembly.

Figure 1.2 shows the cyclo gear axis with one sun and three planet gears that timely drive the three eccentrics and cyclo disks. Also revealed in the two drawings, Figure 1.1 and 1.2, are three hollow eccentric shafts, one hollow center shaft with sun gear, and three additional passageways in the containing

flange. There are also six threaded fastening holes on each side of the axis for tying this axis to other machine units. There are twelve taped holes in the ID-Gear housing for fastening the high torque flange to other machine parts. In total, the drawings reveal seven hollow channels in the patented cyclo gear axis design.

Figure 1.3 shows part numbers 4, 5, and 6 in cut view. This design allows the hallow channels that are an integral part of these inventions.

Figure 1.4 shows, in cut view, the assembled parts numbers 2, 3, 7, 8, 9 as they are being inserted into part number 1, which is the gear housing shown in Figures 1.5 and 1.6.

Figure 1.7 is identical to Figure 1.1. This Figure is shown side by side with Figure 1.8 to depict the three cyclo disk engagements that ensure equalized force distribution.

For clarity and simplification, Figure 1.6 shows a one cyclo disk engagement.

Table 2 clarifies the cyclo gear relations and symbols. In this cyclo gear system, the gear teeth are round. The center of each tooth lays on the gear's true rollup diameter as shown in Figure 1.9.

- *D1* is the roll diameter of the cyclo gear.
- *D2* is the roll diameter of the cyclo disk.
- *Z1* stands for the amount of cyclo gear teeth (always a whole integer).
- *Z2* stands for the number of cyclo disk teeth and is one less than Z1 here.
- *R* stands for the tooth radius.
- *r* stands for the arc – radius, generated by tangent of R, R, and D2.
- *O* stands for the offset of the eccentric and is dimensioned as O= R/2.
- *e* stands for the angular index of the offset and here is 0, 120, 240 degrees.

Variable load deflections do influence cyclo gear engagements. However, with three disks in constant engagement and zero play and backlash between eccentrics and cyclo teeth, motor and encoder, a vibration induced by the servo and cyclo gears is not possible. The vibrations shown in Charts 1 and 2, therefore, are not possible with these cyclo axes designs. This patented cyclo gear axis velocity Chart is a straight parallel line to the time "t" line.

Table 3 shows samples of cyclo relations starting with only three teeth up to 61 teeth. Gear ratios up to 500/1 are recommended with the Figure 2 type cyclo design and 1500/1 with fore-set planetary gear as in the Figure 1 type cyclo design.

Figure 1.11 shows how this geometrical system fits together and how easily it can be expanded. It is an economic and ridged high torque system that fits low, medium, and high lot manufacturing technologies such as casting, powder metal pressing, stamping, and CNC manufacturing.

In Figure 2, the basic cyclo torque multipliers are drawn with one, two, and three wave disks. There is only one centered eccentric driving shaft. But there are six hollow drive-out pins with sleeves kept between in the drive-out flanges. The hub-axis, bearing and housing arrangement are improvements here. The cyclo gear and cyclo disks and their relations are identical to the one shown under Figure 1. The expandability of the cyclo axes design is featured here again.

The radial eccentric index (e) is given at 360 degrees divided by the numbers of the disks. For instance, two disks are 180 degrees apart and 3 disks are 120 degrees apart.

The schematic in Figure 3 shows one add-on harmonic damping filter for the analog electronic velocity feedback loop. It is desirable to increase the life of servo components by reducing or eliminating harmonic vibrations. This filter is very easy to apply and is effective in improving life and performance of machine tools and cyclo gears. It filters and reduces vibration.

The results of the frequency shift filter of Figure 3 schematics are shown as trace curves in Figure 4. Considering any velocity vibrations, the high amplitude spikes will be countered by attenuating and delaying the velocity signal to work against the periodic movements. This is called oscillation and velocity vibration damping by electronic means. This is an active countermeasure and is part of these inventions.

Figure 5 shows the schematics of the battery power back-up transistor to transistor logic with two-channel single disk quadrupling encoder up/down counters. It will replace what is currently in use which is an assortment of different geared disk encoders with a multitude of counters. With this new invention which is a constantly "ON", low-powered (infrared) battery-backed encoder system, the rotation position is always known even when the main power is turned off, lost, or interrupted, as long as the battery is powering the encoder. The battery power should supply the encoder and counter for a minimum of five years. A low battery voltage interlock is mandatory.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

The cyclo torque multipliers and controls are used in robotics manipulators, tooling, and production machines and in many industries. They are basically powered and driven by electric or hydraulic servo motors for doing work, such as positioning and flexible programming to do adaptive moves and locating.

Since the invention of the wheel, man has unsuccessfully tried to understand and document all important geometrical relations of the cyclo drive. The inventor of this patent has achieved the simplification of the cyclo gear with all the necessary geometrical relations as shown in Table 2. Important features are the depths of the gear contacts, overall engagement and tooth spacing.

The cyclo module is the radius of the cyclo tooth or roller radius. The gear pitch diameter is the cyclo-module times the number of teeth and it represents the roll-up diameter. The arc roller spacing is also the diametrical pitch. The eccentric offset size "O" is the cyclo module "R" multiplied by 0.5.

Figures 1.1 through 1.8 show how the eccentric shafts, with the bearings and the flanges, together with the cyclo ID-Gear housing, generate high torque. Because of the three cyclo disks and the deep and direct radial contact, the force deflection is neutralized. The shock safety load of this cyclo gear design is very high. There is practically zero load deflection because of the short drive in and drive out shafts which are supported at both ends.

The output torque equals 0.98 x gear-ratios x input torque. This cyclo gear axis efficiency is very high 98%.

The large hollow eccentric shafts make coaxial lead-through possible. The seven hollows allow passing through of cables, shafts, etc. Because of the two bearing supports of the axis system, the radial and axial loads can be much higher as in cantilevered drive-out cyclo systems.

The Figure 3 circuit is a dampening anti-oscillating add-on filter for servo systems velocity signal. Inertia, imbalance, cantilevered drive-out, one and two disks cyclo systems without bearing support, and manufactured imperfection influence rotating shafts, gears, and machine elements. These imperfections quite often show up as vibration and oscillation. Servo drives, because of the feedback, and the phase delays stimulate vibration especially if the servo response is working in a high-gain mode. To minimize or eliminate this problem, the add-on-filter for servo systems was invented.

At Figure 3, the servo correctional signal enters the servo amplifier. Without the servo filter one of the following signals becomes an under-damped signal, as shown in Figure 4. But with the servo frequency and damping filter installed, the Figure 4 signal looks like the critically-damped signal. Connecting the servo Amplifier Output to C1, an added delay will be added and the trace will look like the critical-damped and delayed signal. Without the Figure 3 frequency shift and damping filter, vibrations shown in Chart 1 and 2 are standard. However, the cyclo axes inventions shown here, together with the servo filter and absolute encoder system will perform smoothly for many productive decades.

The Figure 5 schematic shows the invention of the new absolute rotating encoder. The rechargeable battery will power the LED and will emit a light beam. The angular rotation encoder has transparent slot windows that let the light path through or stop it. The light will trigger the photo-transistors ON and OFF, making electric square pulses in the channel A and B. If the Channel "A" pulse is leading, the up-counter is counting with increasing counts. If the Channel "B" pulse is leading, then the down-counter is decreasing the counts. The counts represent an axis or gear positions in machine tools, robots, etc. The shift register allows a computer, for instance, to access the counter data for position verification. The battery is charging when the system is powered. The battery is powering the axis counter at all times. This arrangement constitutes an inexpensive absolute counter. This system reduces the absolute encoder cost noticeably and increases the absolute encoder reliability by a minimum of 1000% because of fewer components in use. The cyclo axes positioning is very reliable and completes the system.

The Claims are:

1. A total design system for planet-type roller gears. This system is also known as cyclo gear torque multiplier. The basic geometrical relationships evolve around the "cyclo module", the radius, "R" of the cyclo tooth. These relations are shown in Table 2. The simplifications and improvements of the cyclo gear axis system, the basic features of these claims, are:

2. A geometric design arrangement for planet type roller gear according to Claim 1 wherein the roller radius has the given relation to the cyclo-module as shown in Figure 1.9.

3. A geometric design arrangement for planet type roller gears to Claim 2 wherein the roller size R, the roll-up diameter D2, provide the three tangent points to generate the tooth cup radius "r" of the cyclo disk as illustrated under Figure 1.9 and Table 2.

4. A geometric design arrangement for planet type roller gears according to Claim 3 wherein the eccentric has a geometric relation to the cyclo module as shown in Figure 1.9 and Table 2.

5. A geometric design arrangement for planet type roller gears according to Claim 4 wherein the wave disk has a geometrical relation to Claim 2 and 3 and Figure 1.9 and Table 2.

6. A geometric design arrangement for planet type roller gears according to Claim 5 wherein three eccentrics are indexed equally around the

center as shown in drawings Figure 1.1, 1.2, 1.8, 1.10.

7. A geometric design arrangement for planet type roller gears according to Claim 6 wherein the number of eccentrics shown are 1, 2, or 3, as drawn in Figure 2.1, 2.2, 2.3.

8. A geometric design arrangement for planet type roller gears according to Claim 7 wherein the eccentrics are spaced to drive the high torque generated by the cyclo gear and wave disks in connection with the containing flanges as shown in Figure 1.1, 1.2.

9. A geometric design arrangement for planet type roller gears according to Claim 8, wherein the two drive-out flanges are driven by the eccentrics play-free bearings as in Figure 1, 2.

10. A geometric design arrangement for planet type roller gears according to Claim 9 wherein flange and housing bearings form a complete unit axis-cyclo-gear-assembly with taped mounting holes, as shown in Figure 1.1, 1.2.

11. A geometric design arrangement for planet type roller gears according to Claim 10 wherein six hollow torque stabilizing bars with sleeves, stabilize the two drive-out flanges as shown in Figures 2.1, 2.2, 2.3.

12. A geometric design arrangement for planet type roller gears according to Claim 11 wherein a pair of deep groove or cross-roller bearing is used to stabilize the high torque flange to the gear housing, as in Figures 1.1, 2.1, 2,2, 2.3, to make the gear assembly an axis or turntable.

13. A geometric design arrangement for planet type roller gears according to Claim 12 wherein all cyclo rollers are reset or hallowed and pinned as shown in Figure 2.1, 2.2, 2.3.

14. A geometric design arrangement for planet type roller gears according to Claims 1 through 13, wherein the rotating position accuracy is further enhanced by controlling its position at any time by adding an absolute shaft encoder to the gear axis drive-in as shown on Figure 5.

15. A geometric design arrangement for planet type roller gears according to Claim 14 wherein a two channel absolute angular encoder with up/down counter is continuously powered to make it an absolute position smart axis as shown in Figure 5.

16. A geometric design arrangement for planet type roller gears according

to Claim15, wherein the analog summing circuits and feedback servo circuit may feed back data misdirecting the summing results and servo action. The Figure 3 frequency and servo filter counteracts extraneous signals and enhances further the productivity and performance of the cyclo torque multiplier and cyclo gear axis as shown in Figure 4.